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SUMMARY

Parametric tests were conducted in a high-speed bearing tester on a 118-millimeter-bore roller bearing with a round outer ring. The bearing was manufactured from consumable-electrode-vacuum-melted AISI M-50 steel with a room-temperature Rockwell C hardness of 60. Test parameters were radial loads of 2200, 4400, 6700, and 8900 newtons (500, 1000, 1500, and 2000 lb) and nominal speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature (for both the lubricating and the cooling oil) was 366 K (200° F). Oil was supplied through the inner ring for lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 10.2×10^{-3} cubic meter per minute (0.5 to 2.7 gal/min). Outer-ring cooling flow rates were 0 to 4.3×10^{-3} cubic meter per minute (0 to 1.13 gal/min). The lubricant, neopentylpolyol (tetra) ester, met the MIL-L-23699 specifications.

The 118-millimeter-bore roller bearing ran successfully to 3.0×10^{6} DN with radial loads of 2200 to 8900 newtons (500 to 2000 lb) and no outer-ring cooling. At a maximum total flow rate to the inner ring of 9.5×10^{-3} cubic meter per minute (2.5 gal/min) and no outer-ring cooling, the maximum bearing temperature was 466 K (380° F).

No cage slip occurred at 3.0×10^{6} DN primarily because of the tight clearance or even slight interference within the bearing at this high speed. Bearing temperatures varied inversely with cage slip for all conditions. There was no effect of load on either bearing temperature or cage slip as load was varied from 2200 to 8900 newtons (500 to 2000 lb) over a speed range of 10 000 to 25 500 rpm.

Cooling the outer ring decreased its temperature but increased the inner-ring temperature. With no outer-ring cooling, bearing temperatures decreased with increasing total oil flow rate to the inner ring. Outer-ring temperatures were always higher than inner-ring temperatures over the shaft speed range of 10 000 to 25 500 rpm. Heat rejected to the lubricant (power loss within bearing) increased with both shaft speed and total oil flow rate to the inner ring.

The test bearing, except for the cage, showed no appreciable wear on any of its elements after running 111 hours at speeds from 10 000 to 25 500 rpm. One area on the silver plate of the cage's land riding surface showed some wear, which was probably due to cage unbalance.

INTRODUCTION

Advanced aircraft turbine engines in the 1990's will require rolling-element bearings that operate at speeds to 3.0×10^6 DN. (DN is defined as the speed of the bearing in rpm multiplied by the bearing bore in millimeters.) At these extreme conditions, bearing lubrication through the inner ring is mandatory. References 1 and 2 report that ball bearings operating at 3.0×10^6 DN can run successfully with this method of lubrication. With the exception of reference 3, most experimental research has been on high-speed ball bearings, with very little on high-speed roller bearings. Of the innumerable parameters that affect bearing performance, only a few can be investigated in any one experimental study. Some basic parameters that can affect bearing performance are lubricant flow to the inner ring, inner-ring cooling, outer-ring cooling, oil-inlet temperature, bearing load, and inner-ring (shaft) speed.

One possible problem inherent in rolling element bearings operating at high DN values under light loads is slip. Slip is the relative difference in velocity between components that should be rolling, that is, that should have the same surface velocities (ref. 4). Slip occurs when the radial load is insufficient to develop a tractional drive force to overcome friction and drag between the rotating raceway and the rolling bearing elements. Slip may not result in surface damage and can even be advantageous, as discussed in the section Effect of Cage Slip on Bearing Temperature, as long as there is no breakdown of the oil film that separates the slipping elements. Surface damage during slip is called skid damage (ref. 4). Since slip can lead to skidding, which is a prelude to bearing failure, a number of methods have been employed to prevent slip. Decreasing internal clearance or using out-of-round designs of the outer ring, which in effect increases the load on the roller bearing by preloading it (ref. 5), are two such methods.

The investigation reported herein was conducted to produce experimental data on roller bearings at speeds to 3.0×10^6 DN. The objective was to determine the parametric effects of speed and load on bearing performance. The tests were conducted in a high-speed bearing tester on a conventional 118-millimeter-bore roller bearing that was designed for the main shaft of an aircraft turbojet engine and that had been modified for lubrication through the inner ring. A round outer ring was used in this investigation. Appropriate selection of fits and bearing free-state clearance resulted in a bearing assembly that had an operating clearance (or slight interference) sufficient to prevent slip at maximum operating conditions of 3.0×10^6 DN. The bearing was manufactured from consumable-electrode-vacuum-melted AISI M-50 steel. The room-temperature hardness was Rockwell C-60. Test parameters were radial loads of 2200, 4400, 6700, and 8900 newtons (500, 1000, 1500, and 2000 lb) and nominal shaft speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature (for both the lubricating and the cooling oil) was 366 K (200° F). Oil was supplied through the inner ring for both

lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 10.2×10^{-3} cubic meter per minute (0.5 to 2.7 gal/min). Outer-ring cooling rates were 0 to 4.3×10^{-3} cubic meter per minute (0 to 1.13 gal/min). The lubricant, neopentylpolyol (tetra) ester, met the MIL-L-23699 specifications.

APPARATUS AND PROCEDURE

A cross-sectional drawing of the high-speed, roller-bearing test rig is shown in figure 1. The test shaft was mounted vertically and was supported radially by two roller bearings equidistant from the 118-millimeter-bore test roller bearing. The outer ring of the test bearing was assembled into a cylindrical housing suspended by rods that protruded from the top cover of the test rig. The housing could be moved axially to allow centering of the outer ring with the radial load line. The rollers of the test bearing were centered in the outer-ring length of the test bearing by adjusting the shaft vertically. Radial load was applied by a hydraulic cylinder attached through a linkage mechanism to a radial load strap that was wrapped partially around the housing. The test shaft was driven by an air turbine located at the bottom of the main test-shaft assembly.

Measured lubricant flow through an in-line turbine flowmeter provided oil to the test bearing by means of two jets that fed an annular groove adjacent to the bearing (fig. 2). Oil was pumped by centrifugal force through the grooves in the test-bearing bore and through a series of small radial holes to the rolling elements. Those axial grooves in the bearing bore that did not have radial holes allowed oil to flow axially under the ring for inner-ring cooling. Measured cooling oil was supplied through an in-line turbine flowmeter to the outer ring by means of holes and grooves in the bearing housing (figs. 1 and 2).

Shaft speed (inner-ring speed) was measured with a magnetic probe. Cage speed was measured with a magnetic probe on one face of the test-bearing cage and an induction probe on the opposite face. Inner-ring temperature was measured with two spring-loaded thermocouples that projected radially outward through the shaft center at the test-bearing location. Outer-ring temperature was measured with three equally spaced spring-loaded thermocouples in the test-bearing housing, one 15° from the center of the bearing load zone. The temperature of the oil supplied to the test-bearing inner ring was measured near the jets that supplied the annular groove that fed the bearing. The temperature of the oil discharged from the test bearing was measured in an oil collector that had a thermocouple at the bottom of its drain hole (figs. 1 and 2). In this way, very accurate readings of the oil-inlet and -outlet temperatures were obtained.

Test Bearing

The test bearing was an ABEC-5 grade, 118-millimeter-bore roller bearing with an inner land riding cage. The original bearing had an out-of-round (lobed) outer ring for preload; it was replaced by a round outer ring. The bearing contained 28 rollers, 12.6 millimeters (0.4979 in.) in diameter and 14.6 millimeters (0.5734 in.) long. Complete specifications are shown in table I.

The disassembled bearing is shown in figure 3. The bearing design permitted lubrication through the inner ring by means of axial grooves machined in the bore. This technique for high-speed-bearing lubrication has been shown to be most reliable (refs. 1 and 2). Sixteen 0.010-centimeter- (0.025-in.-) diameter holes radiating from the bearing bore to the two circumferential grinding relief grooves in the inner ring formed a flow path for lubrication of the rolling elements. There were 16 radial holes and 31 axial grooves in the bearing bore. Therefore, it was assumed that, for all test speeds, approximately 52 percent of the oil supplied to the inner ring lubricated the bearing and the remainder flowed axially through those grooves that contained no radial holes. The latter flow cooled the inner ring.

Lubricant

The oil used for the parametric studies was a neopentylpolyol (tetra) ester, a type II oil qualified to the MIL-L-23699 specifications as well as to the internal oil specifications of most major aircraft engine producers. The major properties of the oil are presented in table II, and the temperature-viscosity curve is shown in figure 4.

Fits and Clearances

The test-bearing fits and clearances are shown in table III. Because a round outer ring was substituted for the original lobed ring, the bearing was designed with a very small clearance or even a slight interference (preload) at 3.0×10^{6}-DN operating conditions. This would insure minimum cage slip at high speed. A mounted bearing diametral clearance, at zero speed, of 0.086 millimeter (0.0034 in.) was chosen, with a loose outer-ring fit of 0.023 millimeter (0.0009 in.) in its housing. Under these conditions the bearing would gradually lose its initial 0.086-millimeter (0.0034-in.) mounted diametral clearance with increasing speed, as shown in figure 5. The centrifugal growth of the shaft was calculated by the method shown in reference 6. In figure 5, the curve labeled "total clearance loss" was obtained by adding the clearance change due to interference fit to the clearance change due to centrifugal growth of the inner raceway.

The term "diametral clearance" refers to the total free movement of the bearing components in a radial direction. When a shaft speed of 25 500 rpm (3.0×10^6 DN) was reached, more than the total free-state diametral clearance was taken up, as shown by the total-clearance-loss curve in figure 5. This should impose a load on the rollers. However, since the outer ring was assembled 0.023 millimeter (0.0009 in.) loose in the housing, the thermal growth due to high-speed operation expanded the outer ring and to a lesser extent the bearing housing. This resulted in a tight fit in the housing and a 0.023-millimeter (0.0009-in.) gain in diametral clearance. The pinch, or loading, per roller was thus reduced to about 0.0013 millimeter (0.00005 in.) (fig. 5). The effects of temperature, other than the thermal growth of the outer ring and its housing, were neglected in these calculations. Therefore, some deviation from the calculated roller load conditions can be expected.

Test Procedure

The test shaft was brought to the desired speed with a radial load of 2200 newtons (500 lb) on the bearing. When the inner-ring oil-inlet temperature stabilized at 366 K (200° F), the desired total oil flow rate was set. When all bearing temperatures stabilized (after about 20 min), conditions were set for data acquisition. Data were subsequently taken at four loads - 2200, 4400, 6700, and 8900 newtons (500, 1000, 1500, and 2000 lb) - at constant flow rate. This procedure was repeated at four speeds - 10 000, 15 000, 20 000, and 25 500 rpm - at oil flow rates to the inner ring from 1.9×10^{-3} to 10.2×10^{-3} cubic meter per minute (0.5 to 2.7 gal/min). (A running time of at least 20 min was allowed at each condition before data were acquired.) Some of the tests were run with outer-ring cooling rates from 0 to 4.3×10^{-3} cubic meter per minute (0 to 1.13 gal/min). Because of the physical geometry of the inner ring, it was assumed that 52 percent of the total oil flow supplied to it fed the rollers and cage and that the remainder flowed through the inner-ring axial grooves for cooling.

RESULTS AND DISCUSSION

Parametric tests were conducted in a high-speed bearing tester on a 118-millimeter-bore roller bearing with a round outer ring. Test parameters were radial loads of 2200, 4400, 6700, and 8900 newtons (500, 1000, 1500, and 2000 lb) and nominal speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature was 366 K (200° F). Oil was supplied through the inner ring for lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 10.2×10^{-3} cubic meter per minute (0.5 to

2.7 gal/min). Outer-ring cooling flow rates were 0 to 4.3×10^{-3} cubic meter per minute (0 to 1.13 gal/min). The results of this study are presented in this section.

Effect of Load on Bearing Temperature

The effect of load on bearing temperature at four shaft speeds is shown in figure 6. Oil flow to the inner ring was essentially constant at 6×10^{-3} to 7×10^{-3} cubic meter per minute (1.5 to 1.8 gal/min) at a constant temperature of 366 K (200° F). Load was varied from 2200 to 8900 newtons (500 to 2000 lb). Temperatures of both the inner and outer test-bearing rings remained constant as the load was increased at each speed tested. The greatest difference between the inner- and outer-ring temperatures was 20 K (36 deg F) at 25 500 rpm. The outer ring was always at a higher temperature than the inner ring over the speed range tested. No outer-ring cooling was used in these tests, and the highest bearing temperature recorded was 486 K (416° F). Since these changes in load did not have a significant effect on bearing temperature, most of the remaining tests were conducted at the maximum load of 8900 newtons (2000 lb).

Effect of Total Oil Flow to Inner Ring on Bearing Temperature

The effect of the total oil flow to the inner ring on test-bearing temperature is shown in figure 7. No direct outer-ring cooling was used in these tests. Oil flow rate was varied from 1.9×10^{-3} to 10.2×10^{-3} cubic meter per minute (0.5 to 2.7 gal/min). Speed was varied from 10 000 to 25 500 rpm.

Bearing temperature decreased with increasing flow rate for all speeds tested. The outer ring was always at a higher temperature than the inner ring for all test conditions. The reason is primarily that the inner ring was cooled directly and the outer ring was cooled only with oil that had been heated by the bearing elements. Bearing temperatures increased with speed, as expected. The highest temperature recorded during these tests, 501 K (442° F), was the outer-ring temperature at a flow rate of 5.0×10^{-3} cubic meter per minute (1.4 gal/min) and a speed of 25 500 rpm. This is an extremely low flow rate for successful operation of a 3.0×10^{6}-DN bearing. At a more reasonable flow rate of 10.0×10^{-3} cubic meter per minute (2.65 gal/min), the outer-ring temperature with no outer-ring cooling was 464 K (375° F).

The difference between the outer- and inner-ring temperatures ΔT increased with speed, except at 15 000 rpm, which is discussed in the section Effect of Cage Slip on Bearing Temperature. This increase of ΔT with speed is the result of increasing centrifugal forces of the rollers against the outer ring.

Effect of Speed and Lubricant Flow on Power Loss to the Lubricant

Bearing power loss is dissipated in the form of heat transfer to the lubricant by conduction, convection, and radiation to the surrounding environment. To obtain a measure of this heat transfer and power loss within the bearing, the oil-inlet and -outlet temperatures were obtained for all flow conditions. Total heat absorbed by the lubricant was obtained from the standard heat-transfer equation

$$Q_T = MC_p(T_{out} - T_{in})$$

where

Q_T total heat-transfer rate to lubricant, J/min (Btu/min)

M mass flow rate, kg/min (lb/min)

C_p specific heat, J/kg · K (Btu/lb · ^{o}F)

T_{out} oil-outlet temperature, K (^{o}F)

T_{in} oil-inlet temperature, K (^{o}F)

The results of the heat-transfer calculations are shown in figures 8 and 9 as a function of shaft speed and total oil flow rate to the inner ring, respectively. (For convenience, heat-transfer values were converted from J/min to kW.) The heat transfer to the lubricant (power loss within the bearing) increased with speed (fig. 8). The maximum heat transferred was 14.5 kilowatts (840 Btu/min) at a shaft speed of 25 500 rpm and a total oil flow rate of 10.2×10^{-3} cubic meter per minute (2.7 gal/min). Since only 52 percent of the total flow passes through the moving elements of the bearing, the heat rejected to the lubricant is calculated from a flow-rate value equal to 52 percent of 10.2×10^{-3} cubic meter per minute (2.7 gal/min), or 5.3×10^{-3} cubic meter per minute (1.4 gal/min). The heat transfer to the lubricant also increased with total oil flow rate to the inner ring (fig. 9). The rate of increase in heat transfer to the lubricant increased with shaft speed. From these figures it is apparent that a high-speed bearing should not always be operated at high oil flow rates since the power loss in the bearing can become extremely large.

With the 118-millimeter-bore roller bearing operating at 3.0×10^{6} DN and a total oil flow rate to the inner ring of 10.2×10^{-3} cubic meter per minute (2.7 gal/min), figure 8 indicates the total heat transfer to the lubricant is 14.5 kilowatts (840 Btu/min). Under similar conditions with a 124-millimeter-bore roller bearing, reference 3 reports a heat transfer of 14.9 kilowatts (850 Btu/min). Although the agreement in results is good, unfortunately reference 3 does not explain how the heat transfer in the bearing was determined.

Effect of Outer-Ring Cooling on Bearing Temperature

The effect of outer-ring cooling oil flow rate on bearing temperature for various total oil flow rates to the inner ring is shown in figure 10 for shaft speeds of 20 000 and 25 000 rpm. At total oil flow rates to the inner ring of 3.8×10^{-3} to 10.2×10^{-3} cubic meter per minute (1.0 to 2.7 gal/min), the bearing outer-ring temperature decreased as flow to the outer ring increased. The opposite occurred with inner-ring temperature, which actually rose with increased cooling flow rate to the outer ring. One possible explanation is that bearing radial clearance was diminished from an already negative value to a more negative value by cooling the outer ring. At these tighter bearing clearances (preloaded conditions), the inner ring of the bearing is more highly stressed by the rollers than the outer ring (ref. 7), resulting in an increase in inner-ring temperature. The effect of centrifugal force was still present on the outer ring, but the cooling oil flow prevented a temperature increase. As the outer-ring cooling flow rate increased, bearing clearance decreased. This progressive decrease in clearance increased the stress on the inner ring and thus increased its temperature, as shown by all the curves in figure 10. Obviously, this stress was greater than that generated by increasing bearing loading from 2200 to 8900 newtons (500 to 2000 lb), which produced no increase in bearing temperature (fig. 6).

The data from figure 10 have been crossplotted in figure 11 to show the effect of total oil flow rate to the inner ring on bearing temperature for various outer-ring cooling oil flow rates. From these curves it is apparent that there is a practical limit to the amount of cooling flow to the outer ring that will prove beneficial to high-speed bearing operation. An outer-ring cooling flow rate of 1.7×10^{-3} cubic meter per minute (0.45 gal/min) appears to be that practical limit. At higher flow rates, very little additional cooling of the outer ring can be accomplished at 20 000- and 25 500-rpm shaft speeds.

The bearing maximum temperature was limited to 453 K (355° F) at a maximum total oil flow rate of 9.5×10^{-3} cubic meter per minute (2.5 gal/min) and an outer-ring cooling flow rate of 2.5×10^{-3} cubic meter per minute (0.67 gal/min). Even without outer-ring cooling, the maximum temperature was only 466 K (380° F), which occurred at the outer ring (fig. 11(b)).

Effect of Radial Load on Cage Slip

The effect of radial load on cage slip is shown in figure 12 for various total oil flow rates to the inner ring. The outer ring was not directly oil cooled in these experiments. Cage slip C_s is obtained from the following equation:

$$C_s = 100\left[1 - \frac{\omega_c}{\omega_s}(2 + 2S)\right]$$

where

ω_c cage speed

ω_s shaft speed (inner-ring speed)

S radius ratio, r/R

r radius of roller

R radius of inner-ring roller track (raceway)

For each size of rolling-element bearing, there is a ratio of cage speed to shaft speed at which pure rolling occurs. Any value below this ratio indicates cage slip. The ratio for pure rolling R_{ns} is obtained from the following equation:

$$R_{ns} = \frac{1}{2 + 2S}$$

For the 118-millimeter-bore roller bearing, this ratio is 0.4562. Cage slip can be expressed either by using the ratio of cage speed to shaft speed or by converting this to percent cage slip. The latter terminology has been selected for reporting these experimental results. Present opinion (ref. 8) is that a 10-percent cage slip is tolerable. Although this value has not been experimentally verified, it can be used as a basis for determining whether the experimental bearing may be approaching a condition that can lead to bearing failure.

As figure 12 shows, the percent cage slip was not affected by loads from 2200 to 8900 newtons (500 to 2000 lb) at shaft speeds of 10 000, 20 000, and 25 500 rpm. Cage slip generally increased with a decrease in total oil flow rate to the inner ring at 10 000 rpm (fig. 12(a)), but decreased at 20 000 rpm (fig. 12(b)). Cage slip varied from 46.3 to 52 percent at 10 000 rpm. This is very much above the 10 percent allowable for reliable bearing operation. (At 10 000 rpm as much as 0.069-mm (0.0027-in.) clearance could have existed (fig. 5) and thus promoted slip.) The cage slip (1.9 percent) at 20 000 rpm was well below the tolerable level. And there was virtually no slip at 3.0×10^6 DN (25 500 rpm), probably because a very small clearance or even slight interference was present in the bearing at that speed, as indicated earlier.

Effect of Cage Slip on Bearing Temperature

The test-bearing temperature as a function of the total oil flow rate to the inner ring is shown in figure 13 for shaft speeds of 10 000, 15 000, 20 000, and 25 500 rpm. At each data point the percent cage slip is indicated. These curves were generated from data taken at loads of 2200 to 8900 newtons (500 to 2000 lb). Since it has been shown that load has negligible effects on bearing temperature (fig. 6) and cage slip (fig. 12), the variations in bearing temperature shown here should not be the result of load variations. The data were plotted without regard to load. In some instances the data point showing the highest bearing temperature was taken at the lowest load, and conversely.

At 20 000 rpm (fig. 13(c)) cage slip was minimal, with a maximum of 1.59 percent at a total oil flow rate to the inner ring of 10.2×10^{-3} cubic meter per minute (2.7 gal/min). At 25 500 rpm (fig. 13(d)) there was virtually no cage slip because of the tight internal clearance or even slight interference at this speed.

A study of figure 13 reveals that in each set of data shown, the bearing temperature was lowest when the percent cage slip was highest. This is most apparent in figure 13(b), which shows the greatest variation in both cage slip and bearing temperature especially at the low 1.9×10^{-3} -cubic-meter-per-minute (0.5-gal/min) total oil flow rate to the inner ring. The inner-ring temperature at this low oil flow rate was 454 K (357° F) at 0.12-percent cage slip, but the temperature decreased to 401 K (262° F) at 56.42-percent cage slip.

The test bearing at certain lower speeds exhibited an interesting phenomenon, where the temperature readings cycled between two fixed values. The maximum temperature coincided with minimum cage slip, and conversely. This cycling occurred because some internal conditions were continually changing since all external conditions were kept constant. A result of this unstable condition is the wide range of temperature data shown in figure 13(b), especially at a total oil flow rate to the inner ring of 1.9×10^{-3} cubic meter per minute (0.5 gal/min). The cycling phenomenon is more clearly shown in figure 14, for this total oil flow rate, an oil-inlet temperature of 366 K (200° F), and a speed of 15 000 rpm. The bearing was run at these conditions for approximately 98 minutes. Only load was changed, and that did not affect the temperature cycling of the test bearing.

A possible explanation for the cyclic variation in bearing temperature shown in figure 14 is as follows: At the bearing temperatures shown, 406 to 459 K (271° to 366° F) during the 98-minute period of this test, the outer ring expanded to take up the loose 0.023-millimeter (0.0009-in.) room-temperature fit of the outer ring in the housing. Since the housing temperature was not measured, its mean was assumed to be about 14 K (26 deg F) lower than that of the bearing outer ring. At this mean temperature the

housing thermal growth would be sufficient to allow the bearing outer ring to grow 0.023 millimeter (0.0009 in.) and remain tight in the housing.

The effect of four interacting factors on the cycling phenomenon with time is shown in figure 15. The cycling of the inner-ring temperature was measured with a spring-loaded thermocouple pressed against the bearing bore surface. The curve is representative of that shown in figure 14. The outer-ring temperature curve was not included since it had been assumed that the inside diameter of the outer ring would not change appreciably during the bearing temperature cycling. The dashed vertical lines in figure 15 pass through all four curves. The intersections of the dashed lines and the curves show what happened within the test bearing during the cycling phase.

At the point where curve A and line 1 intersect, maximum inner-ring temperature had caused the inner ring to expand. At this point the bearing was under conditions of tight clearance (or interference)(curve B), minimum slip (curve C) because of a more uniform load distribution among the rollers, and hence maximum traction force (curve D). Because slip, and therefore its heat generation, had reached a minimum at this point, the inner ring began to cool. As cooling progressed, the inner ring shrank and bearing clearance increased until conditions shown by the intersection of line 2 were reached. The inner-ring temperature (curve A) as measured at its inner surface (bore) was then at a minimum, but bearing clearance was at a maximum (curve B). The increased clearance at this point restricted the load distribution to fewer rollers and resulted in maximum cage slip (curve C) and minimum traction (curve D). The high surface heat generated by the shearing of the oil film between the rollers and the races during slip was gradually absorbed by the inner ring. There was a time lag before the inner-ring thermocouple at the bearing bore actually showed the peak inner-ring temperature resulting from this high heat generation. As the peak temperature was reached, the inner ring had grown, decreasing clearance and reducing slip and its accompanying heat generation. This reduction of the heat-generating source (slip) again began a downward swing of the inner-ring temperature and another similar cyclic behavior pattern.

The principal causes of the cycling phenomenon are (1) changes in bearing clearance caused by cyclic heat patterns due to alternating slipping and pure rolling; and (2) the time lag involved for the surface heat generated during slip (shearing of the oil film) to be absorbed by the full inner-ring thickness and the rings subsequent growth.

Cycling did not occur, or at least was minimized, at high oil flow rates because slip was average. The bearing was at an equilibrium condition at which heat was taken away by the oil and was not flowing into the bearing rings sufficiently to increase their temperatures substantially.

Bearing Condition After Testing

The 118-millimeter-bore test roller bearing shown disassembled in figure 3 was run for a total of 111 hours. In general, the bearing showed no appreciable wear on any of its elements. All the elements, except the silver-plated cage, had a blue discoloration probably caused by oxidation when the inner ring was assembled on the shaft and during test runs since an inert atmosphere was not used in either case. This discoloration was even apparent in the roller track of the inner and outer ring. The inner-ring bore showed no evidence of the characteristic circumferential lines that would have indicated that the inner ring had loosened and turned on the shaft.

Reference 9 predicts that the inner ring under the assembled conditions of 0.071-millimeter (0.0028-in.) diametral interference fit would begin to loosen on the shaft at 17 000 rpm. If this were true, at 25 500 rpm the inner ring should definitely be loose on the shaft. If, in fact, the inner ring was loose on the shaft, as predicted by reference 9, the inner-ring axial lock plate effectively kept it from rotating. However, because figure 5 indicates a clearance change of 0.0152 millimeter (0.0006 in.) at 25 500 rpm (3.0×10^{6} DN) due to interference fit, the inner ring must still be tight on the shaft.

Figure 16 shows typical concentric burnished rings on the ends of two test-bearing rollers. The outer rings originate from the flanges, and the smaller centrally located rings originate from cage-pocket rubbing. Upon close observation these rings appeared to be not wear marks but simply burnished areas. The characteristic failure mode that identifies roller skew is rapid eccentric wear on the end surfaces of a roller (ref. 3). Such wear was not apparent on any of the rollers in this investigation.

Figure 17 shows wear through the silver plate on the locating surface of the cage. Wear occurred in only one area of the inner-cage riding surface probably because of cage unbalance. Cage unbalance after silver plating did not exceed 3 gram-centimeters at 500 rpm. After two small screws were installed on one side of the cage (180° apart) to afford a larger signal output for the cage-speed magnetic probe, the cage was not rebalanced. The mating inner-ring cage riding surfaces were exceptionally free of wear marks. There was some evidence of slight wear in some of the cage pockets, but it did not extend through the silver plate.

SUMMARY OF RESULTS

Parametric tests were conducted in a high-speed, high-temperature bearing tester with a 118-millimeter-bore roller bearing. The bearing was manufactured from consumable-electrode-vacuum-melted (CVM) AISI M-50 steel with a room-temperature Rockwell C hardness of 60. Test parameters were radial loads of 2200, 4400, 6700,

8900 newtons (500, 1000, 1500, and 2000 lb) and nominal speeds of 10 000, 15 000, 20 000, and 25 500 rpm. The oil-inlet temperature was 366 K (200° F). Oil was supplied through the inner ring for lubrication and inner-ring cooling at flow rates from 1.9×10^{-3} to 10.2×10^{-3} cubic meter per minute (0.5 to 2.7 gal/min). Fifty-two percent of the flow was assumed to be directed through radial holes into the bearing rolling elements for lubrication, and the remainder was assumed to flow through axial grooves in the inner-ring bore for inner-ring cooling. Outer-ring cooling oil flow rates were 0 to 4.3×10^{-3} cubic meter per minute (0 to 1.13 gal/min). The lubricant was a neopentylpolyol (tetra) ester that met the MIL-L-23699 specification. The following results were obtained:

1. The 118-millimeter-bore test bearing with a round outer ring ran successfully at 3.0×10^{6} DN with radial loads from 2200 to 8900 newtons (500 to 2000 lb). At maximum total flow rates of 9.5×10^{-3} cubic meter per minute (2.5 gal/min) to the inner ring and 2.5×10^{-3} cubic meter per minute (0.67 gal/min) to the outer ring and 8900-newton (2000-lb) load, the maximum bearing temperature was 453 K (355° F). Under the same conditions without outer-ring cooling, the maximum bearing temperature was 466 K (380° F).

2. No cage slip occurred at the 3.0×10^{6}-DN operating level probably because of the very tight clearance or even slight interference within the bearing at high speeds.

3. Cage slip and bearing temperature were not affected by loads from 2200 to 8900 newtons (500 to 2000 lb) at shaft speeds from 10 000 to 25 500 rpm.

4. Bearing temperature varied inversely with cage slip for all test conditions.

5. Heat transfer to the lubricant (power loss within the bearing) increased with both speed and total oil flow rate to the inner ring. The rate of heat transfer increased with shaft speed.

6. Bearing temperature decreased with increasing total oil flow rate to the inner ring for speeds from 10 000 to 25 500 rpm with no outer-ring cooling. Outer-ring temperatures were always higher than inner-ring temperatures. Cooling the outer ring decreased its temperature but increased the inner-ring temperature.

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TABLE I. - TEST-BEARING SPECIFICATIONS

Bearing bore, mm (in.)	[a]118 (4.6454)
Bearing inner-ring, outer-ring, and roller material	CVM M-50 steel
Cage material	Silver-plated AM56414 steel
Number of axial grooves in bearing bore	31
Number of 0.010-cm (0.025-in.) diam, radial, lubricant feed-holes in inner ring	16
Oil flow to bearing inner ring for lubrication, percent of total	[b]52
Oil flow to bearing inner ring for inner-ring cooling, percent of total	[b]48
Number of rollers	28
Roller diameter, mm (in.)	[a]12.647/12.649 (0.4979/0.4980)
Roller length, mm (in.)	[a]14.562/14.564 (0.5733/0.5734)
Roller end clearance in raceway, mm (in.)	[a]0.033 (0.0013)
Roller axial pocket clearance in cage, mm (in.)	[a]0.272 (0.0107)
Roller tangential pocket clearance in cage, mm (in.)	[a]0.218 (0.0086)
Bearing pitch diameter, mm (in.)	145.00 (5.70)

[a]Measured dimensions before testing.
[b]Assumed value.

TABLE II. - PROPERTIES OF TETRAESTER LUBRICANTS

Additives	Corrosion, oxidation, wear, and foam inhibitors
Kinematic viscosity, cS[a] at -	
311 K (100° F)	28.5
372 K (210° F)	5.22
477 K (400° F)	1.31
Flashpoint, K ($^\circ$F)	533 (500)
Autogenous ignition temperature, K ($^\circ$F)	694 (800)
Pourpoint, K ($^\circ$F)	214 (-75)
Volatility (6.5 hr at 477 K (400° F)), wt %	3.2
Specific heat at 372 K (210° F), J/kg · K (Btu/lb · $^\circ$F)	2140 (0.493)
Thermal conductivity at 372 K (210° F), J/m · sec · K (Btu/hr · ft · $^\circ$F)	0.15 (0.088)
Specific gravity at 372 K (210° F)	0.931

[a]10^{-4} Stoke = 1 m^2/sec.

TABLE III. - TEST-BEARING FITS AND CLEARANCES

Test-bearing diametral clearance (free state), mm (in.)	0.122 (0.0048)
Bearing inner-ring assembled interference fit on shaft (diametral), mm (in.)	0.071 (0.0028)
Bearing outer-ring fit in housing (loose), mm (in.)	0.023 (0.0009)
Test-bearing assembled clearance (diametral) at zero rpm, mm (in.)	0.086 (0.0034)
Test-bearing calculated diametral clearance at 25 500 rpm (3.0×10^6 DN), mm (in.)	[a]-0.025 (-0.0010)
Clearance gained from thermal expansion of outer ring at 25 500 rpm (3.0×10^6 DN), mm (in.)	0.023 (0.0009)
Resultant interference fit on each roller at 25 500 rpm (3.0×10^6 DN), mm (in.)	0.0013 (0.00005)

[a]Negative clearance indicates interference at this point.

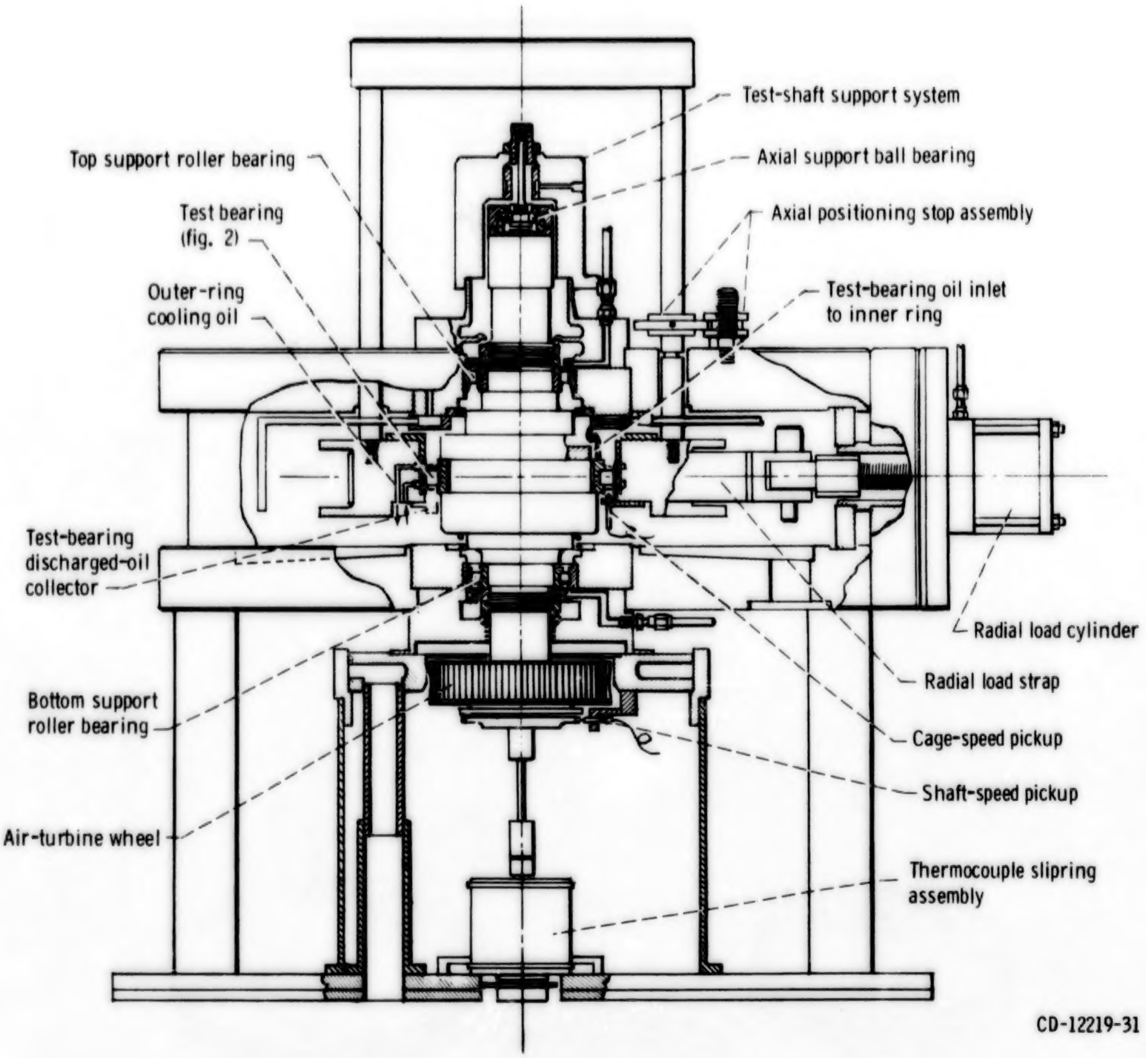

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Figure 1. - High-speed roller bearing test rig.

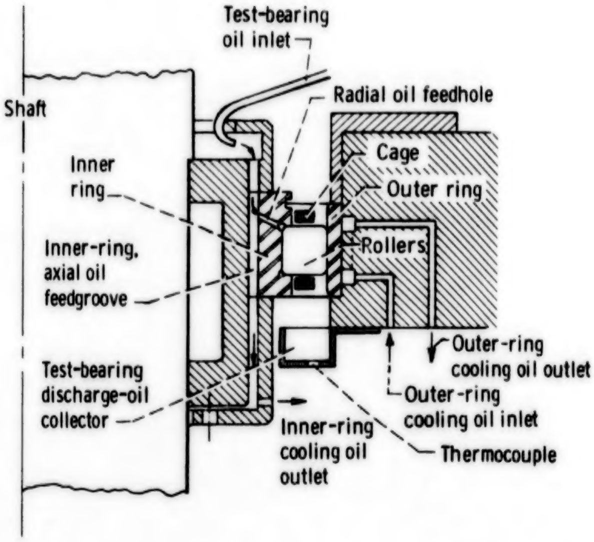


Figure 2. - Test-bearing lubrication and cooling.

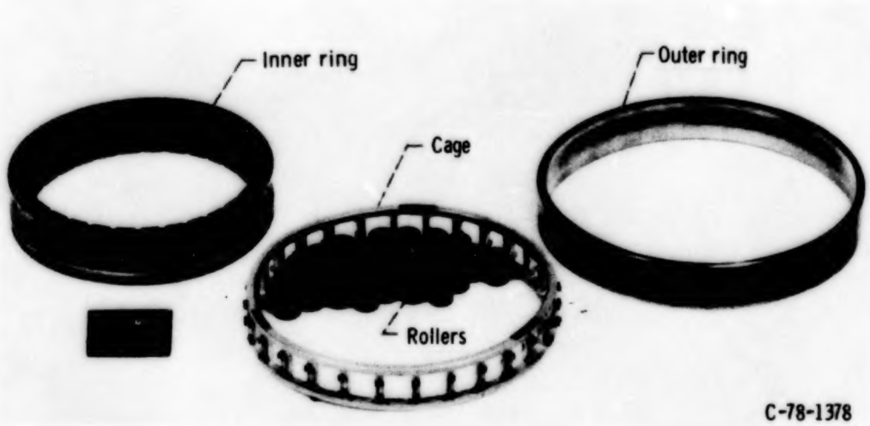


Figure 3. - 118-Millimeter-bore, high-speed roller bearing after 111 hours of running. Radial load, 2200 to 8900 newtons (500 to 2000 lb); shaft speed, 10 000 to 25 500 rpm (1.2×10^6 to 3.0×10^6 DN); maximum bearing temperature, 500 K (441° F).

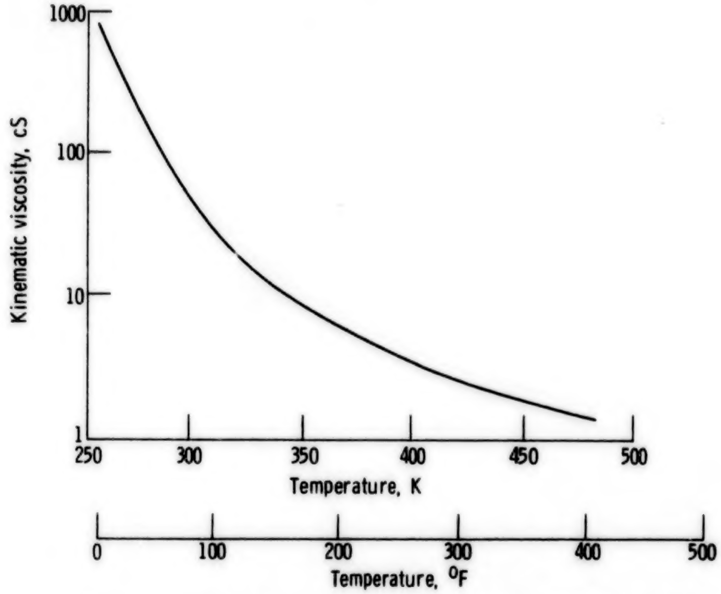


Figure 4. - Effect of temperature on viscosity of neopentylpolyol (type II) lubricant.

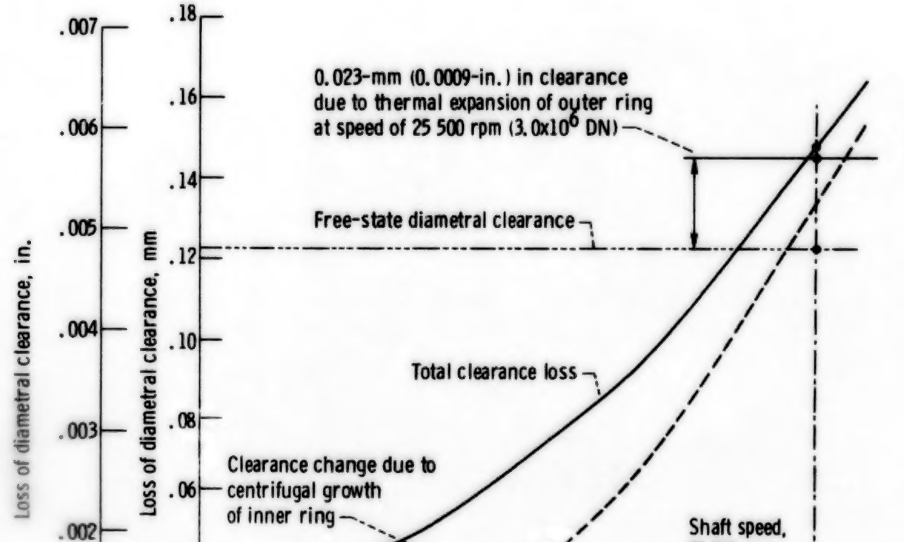


Figure 5. - Effect of shaft speed on dimensional characteristics of test bearing. Bearing inner-ring assembled interference fit on shaft (diametral), 0.071 millimeter (0.0028 in.) at room temperature; free-state diametral clearance, 0.122 millimeter (0.0048 in.); outer-ring fit (loose) in housing, 0.023 millimeter (0.0009 in.) at room temperature.

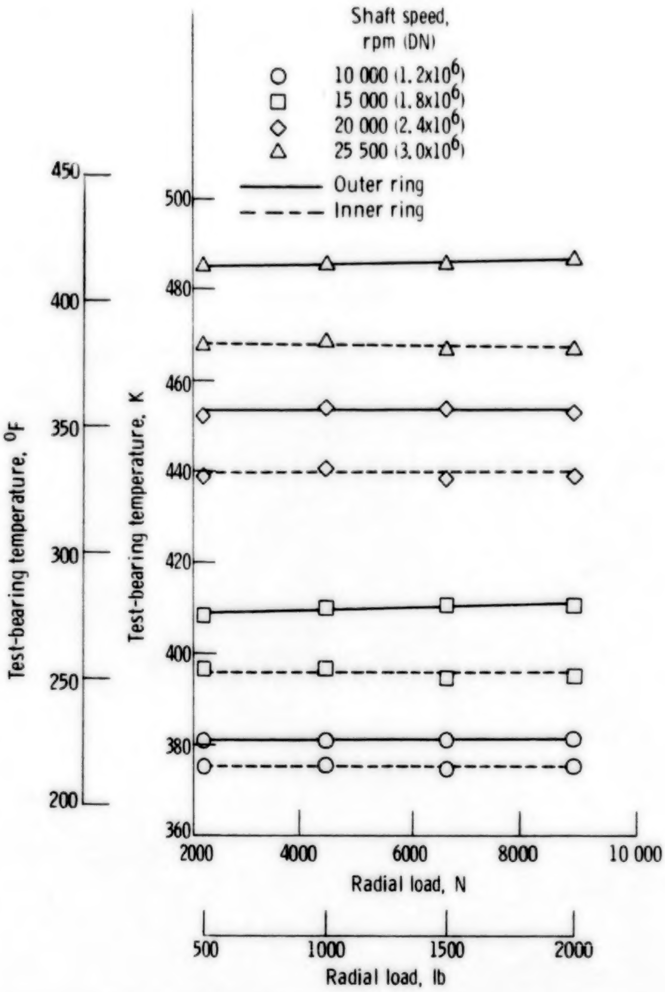


Figure 6. - Effect of load on bearing temperature for various shaft speeds. Total oil flow rate to inner ring, 6.0×10^{-3} to 7.0×10^{-3} cubic meter per minute (1.5 to 1.8 gal/min); oil-inlet temperature, 366 K (200° F).

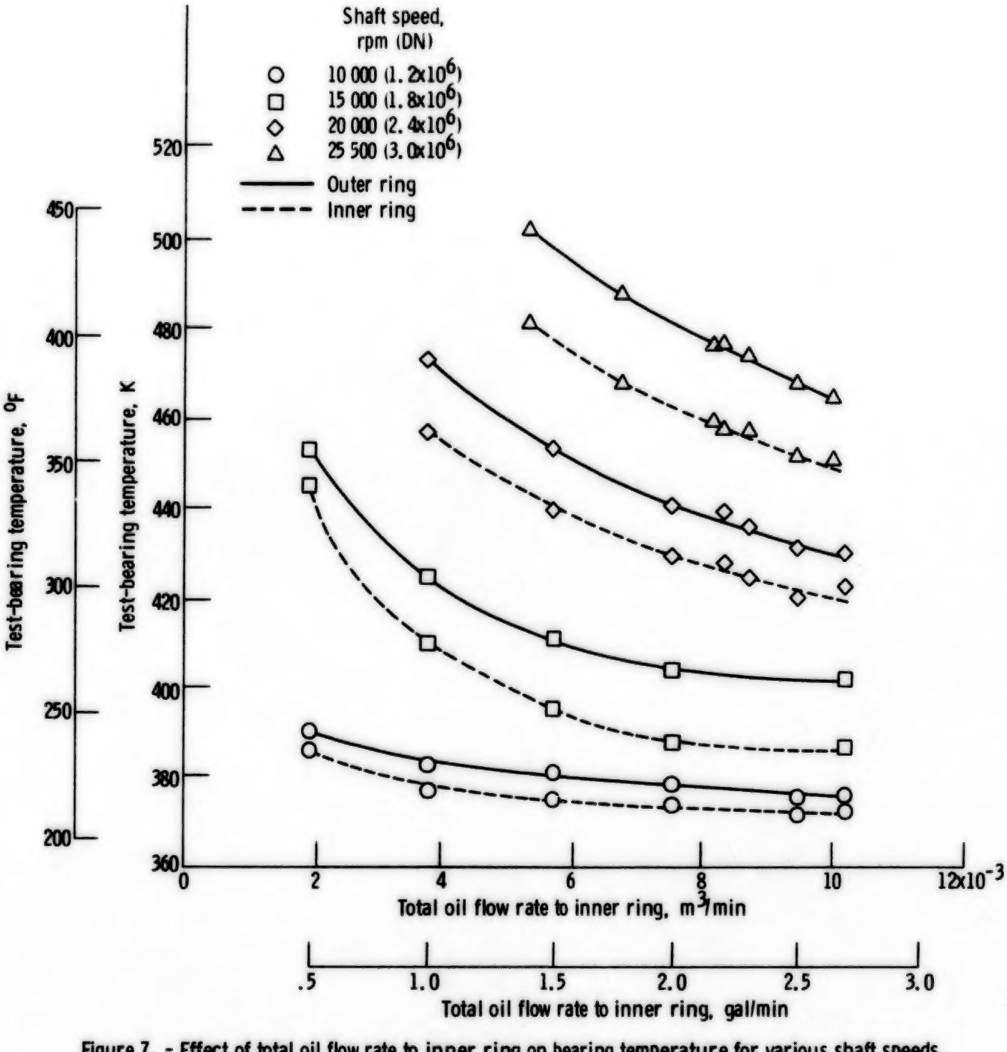


Figure 7. - Effect of total oil flow rate to inner ring on bearing temperature for various shaft speeds. Oil-inlet temperature, 366 K (200° F); radial load, 8900 newtons (2000 lb).

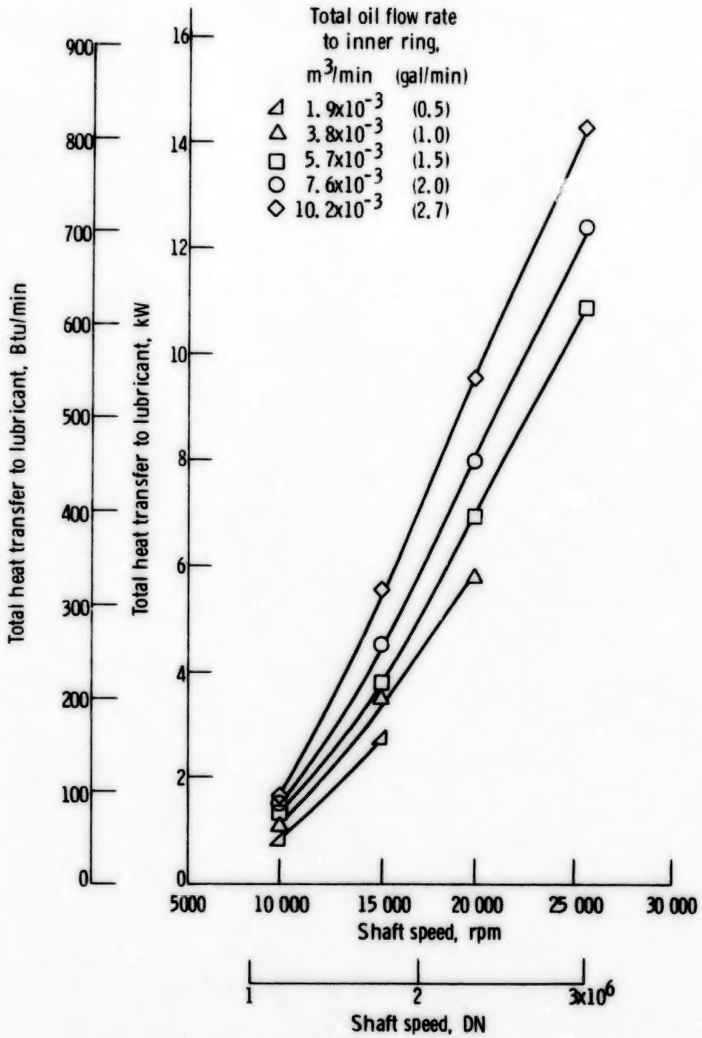


Figure 8. - Effect of shaft speed on heat transfer to lubricant for various total oil flow rates to inner ring. Oil-inlet temperature, 366 K (200° F); radial load, 8900 newtons (2000 lb).

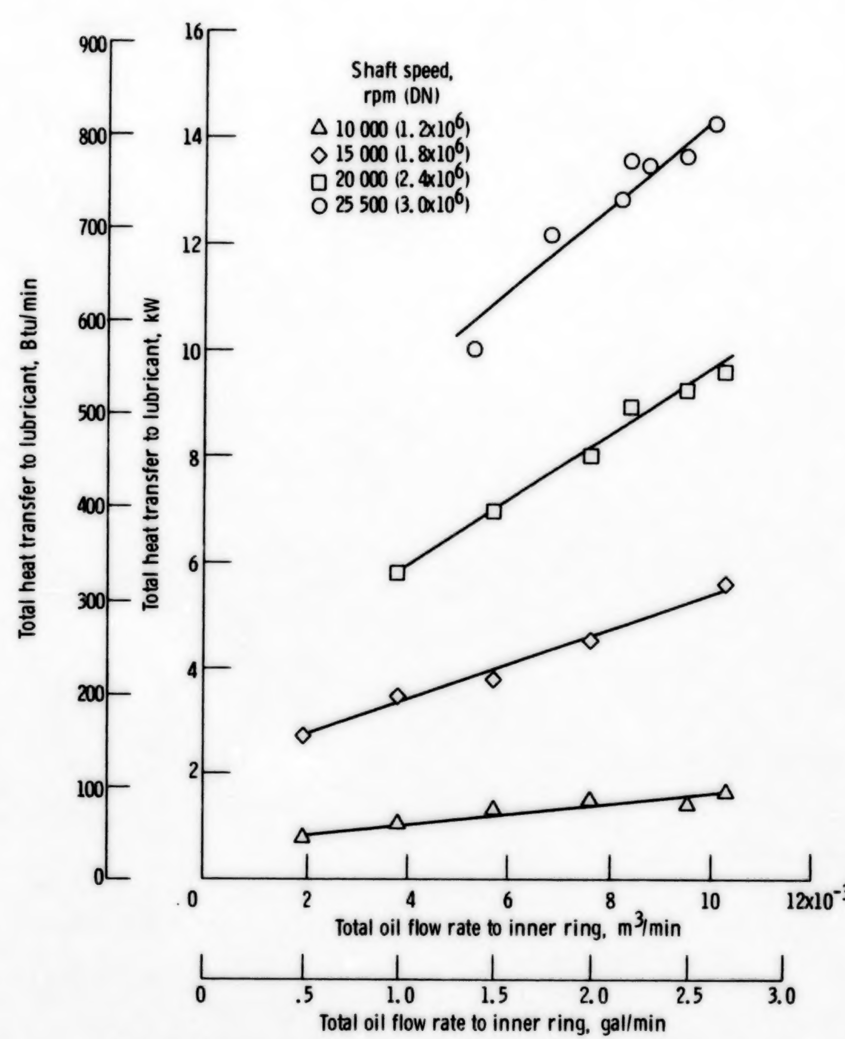


Figure 9. - Effect of total oil flow rate to inner ring on heat transfer to lubricant for various shaft speeds. Oil-inlet temperature, 366 K (200° F); radial load, 8900 newtons (2000 lb).

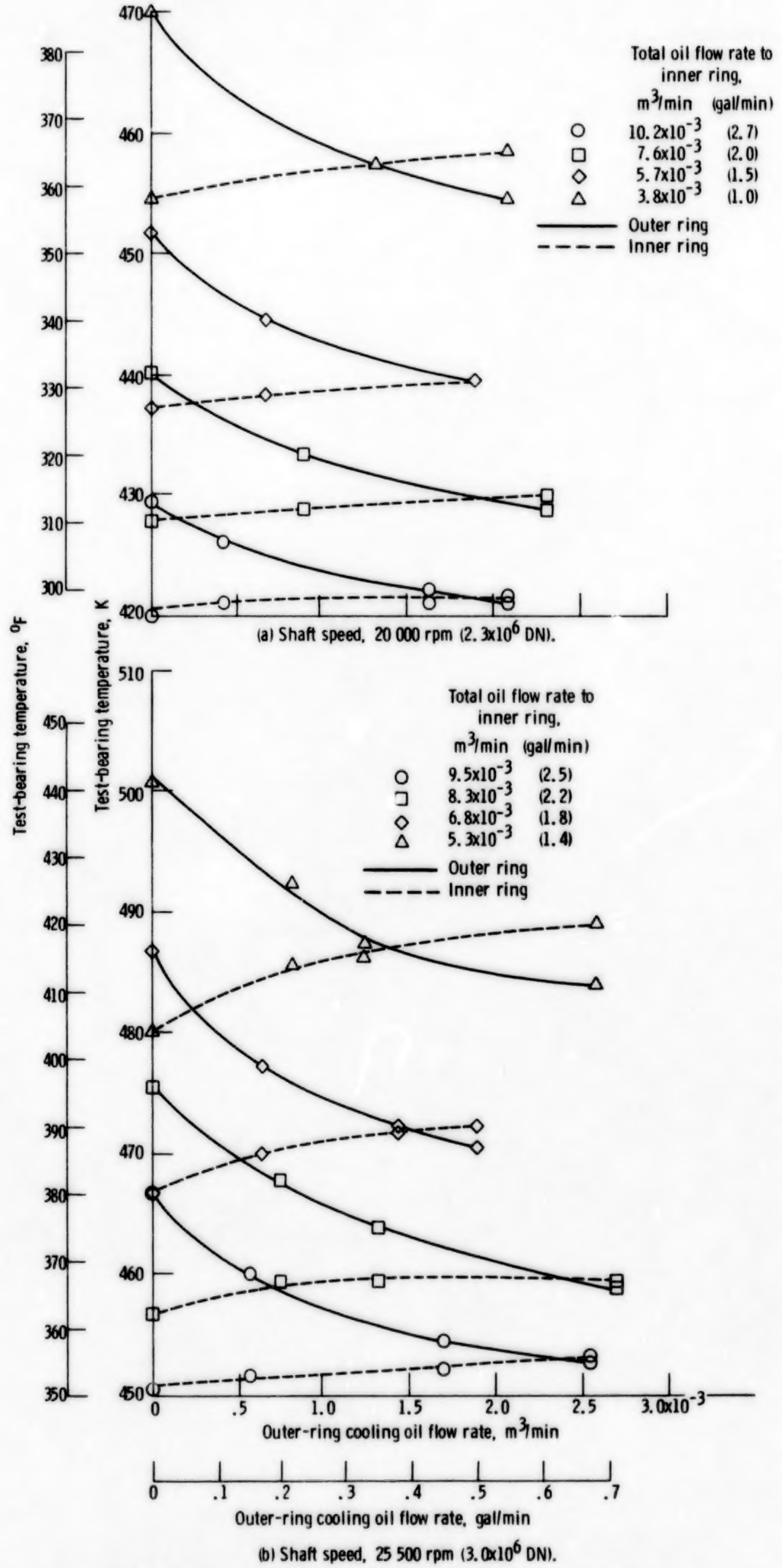
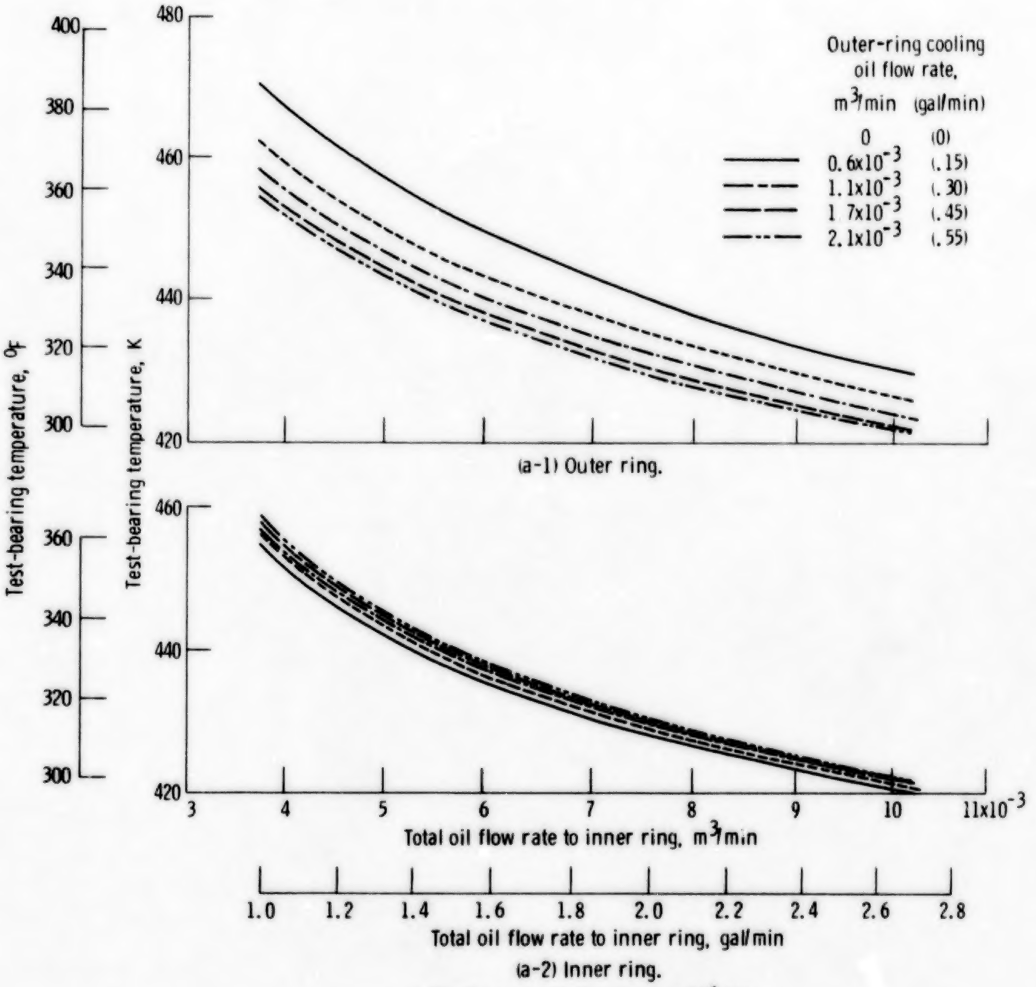
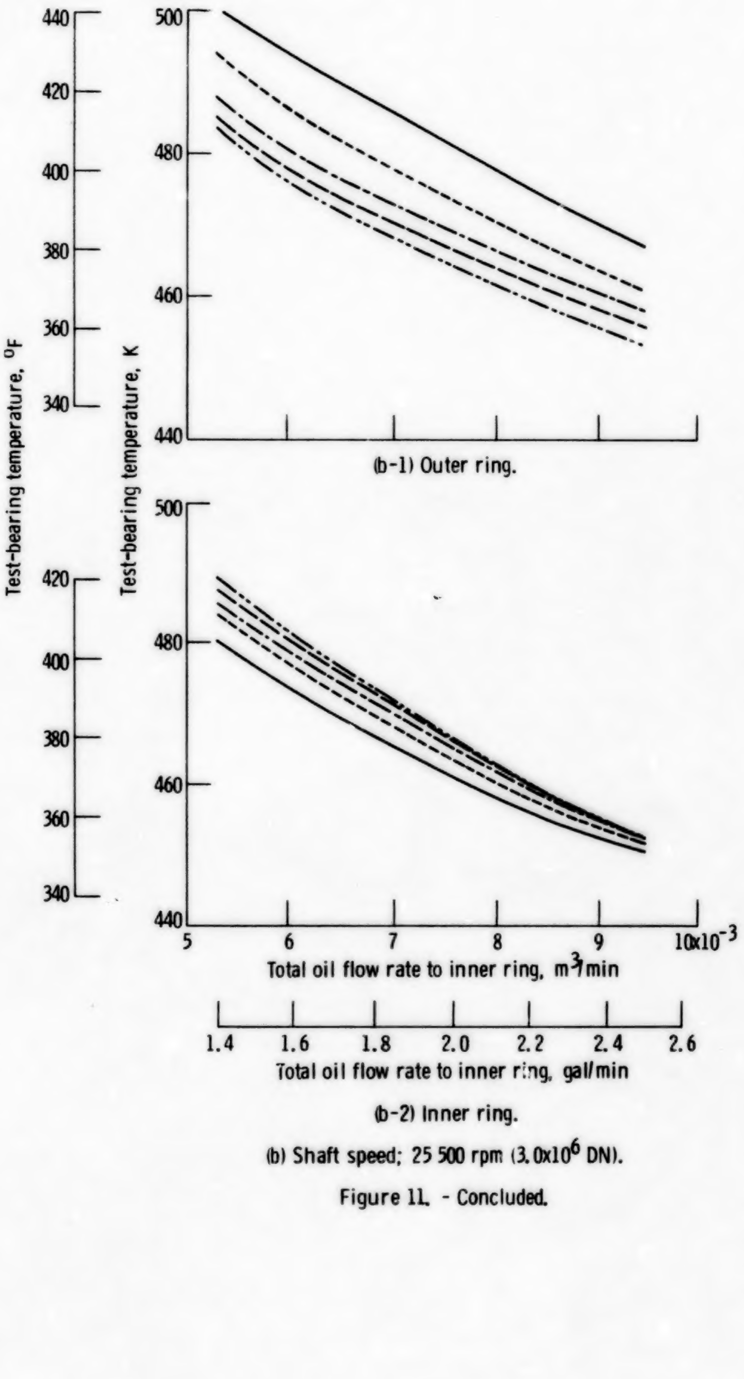


Figure 10. - Effect of outer-ring cooling oil flow rate on bearing temperature for various total oil flow rates to inner ring. Oil-inlet temperature (for both lubricating and cooling oil), 366 K (200° F); radial load, 8900 newtons (2000 lb).



(a) Shaft speed, 20 000 rpm (2.3x10^6 DN).

Figure 11. - Effect of total oil flow rate to inner ring on bearing temperature for various outer-ring cooling oil flow rates. Oil-inlet temperature, 366 K (200° F) for both lubricating and cooling oil; radial load, 8900 newtons (2000 lb).



(b) Shaft speed; 25 500 rpm (3.0×10^6 DN).

Figure 11. - Concluded.

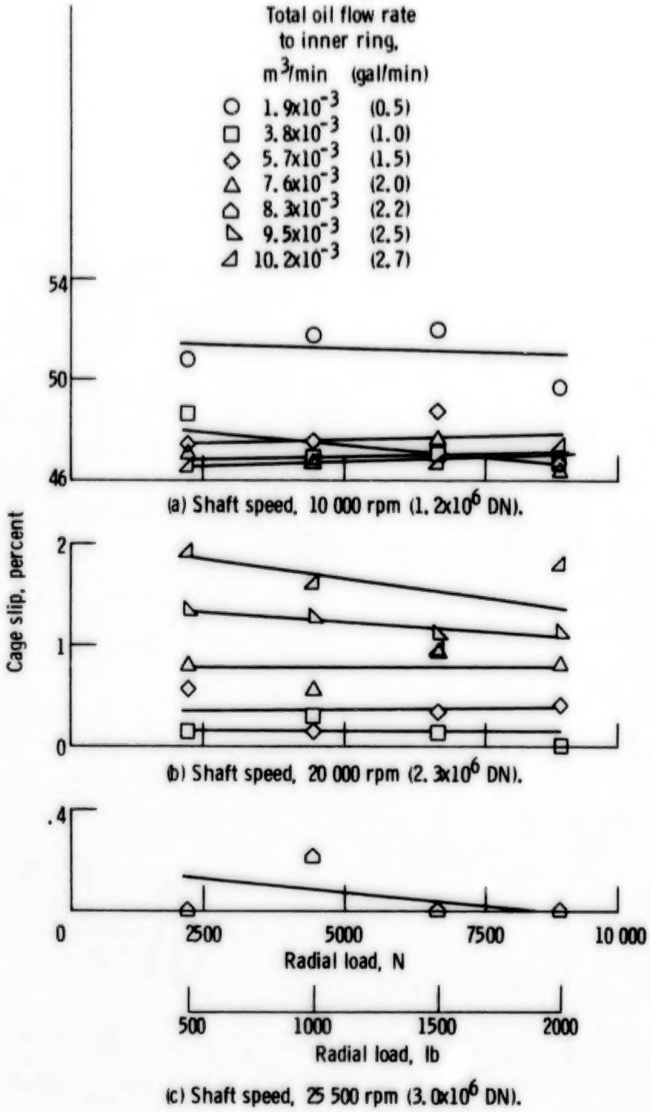


Figure 12. - Effect of radial load on cage slip for various total oil flow rates to inner ring. Oil-inlet temperature, 366 K (200° F).

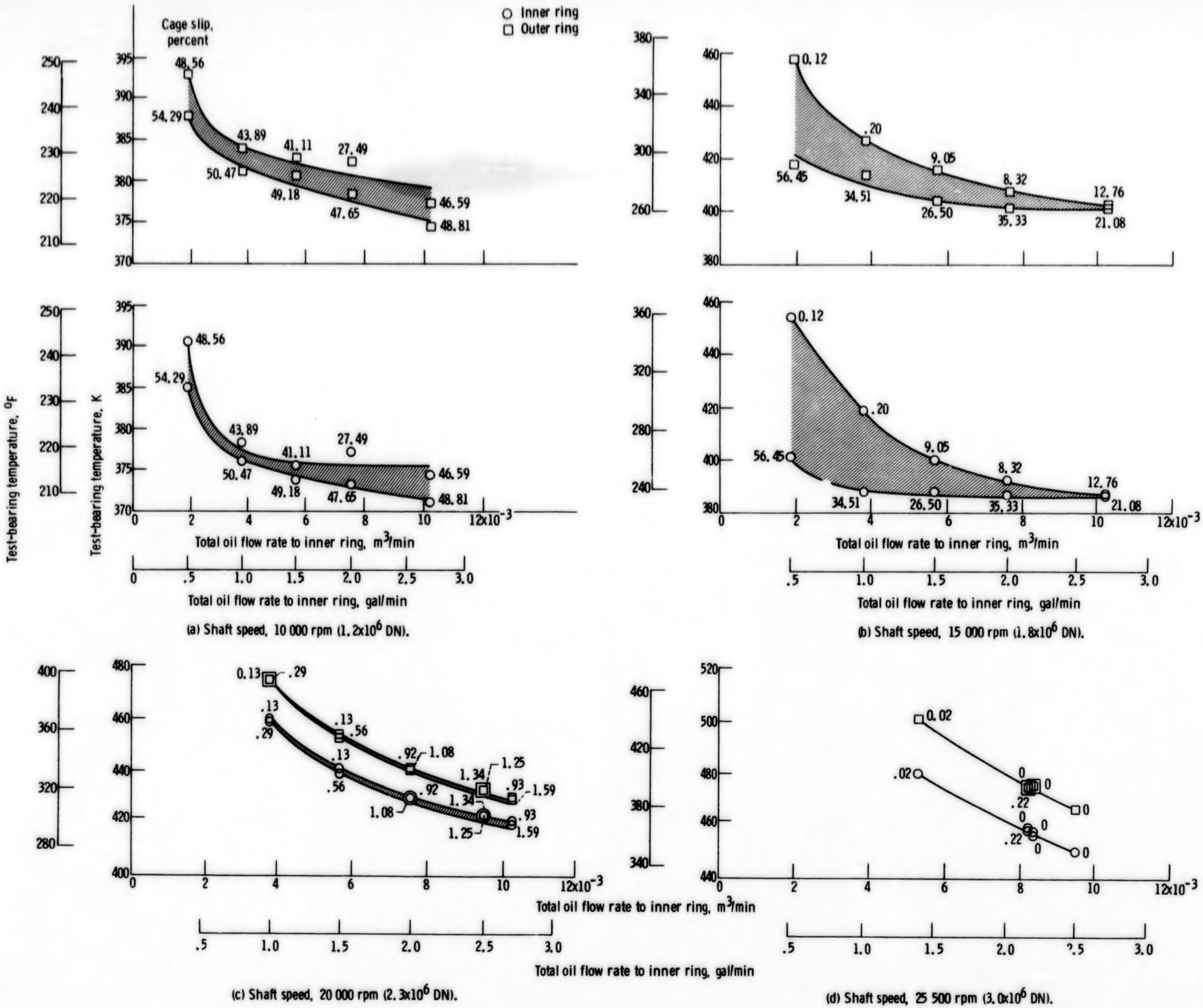


Figure 13. - Effect of cage slip on bearing temperature at various total oil flow rates to inner ring. Oil-inlet temperatures, 366 K (200° F); radial load, varied from 2200 to 8900 newtons (500 to 2000 lb).

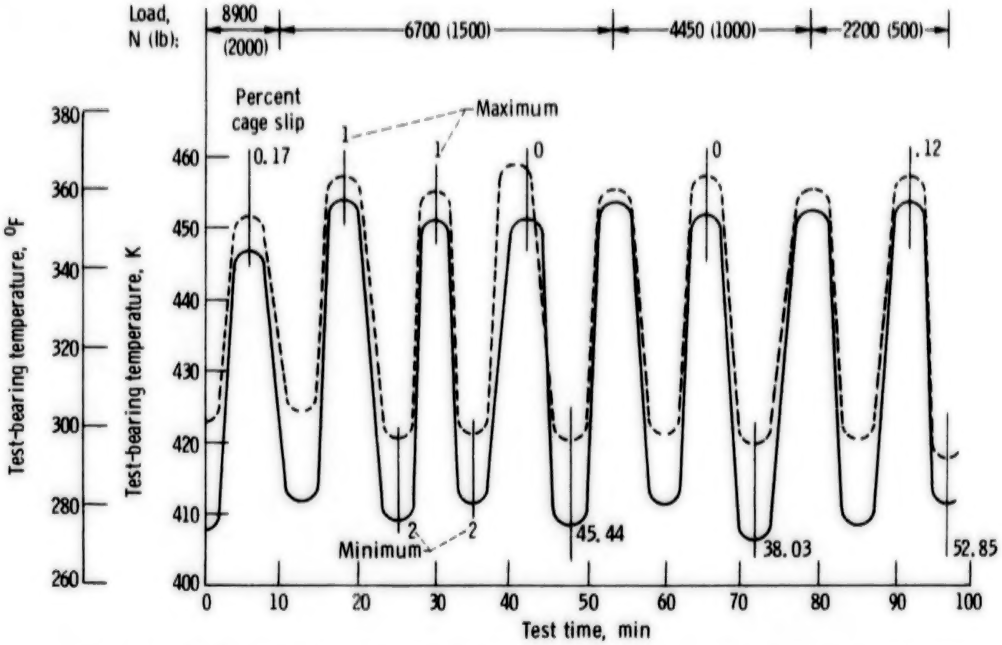


Figure 14. - Effect of cage slip on bearing temperature at various loads. Total oil flow rate to inner ring, 1.9×10^{-3} cubic meter per minute (0.5 gal/min); oil-inlet temperature, 366 K (200° F); shaft speed, 15 000 rpm (1.8×10^{6} DN).

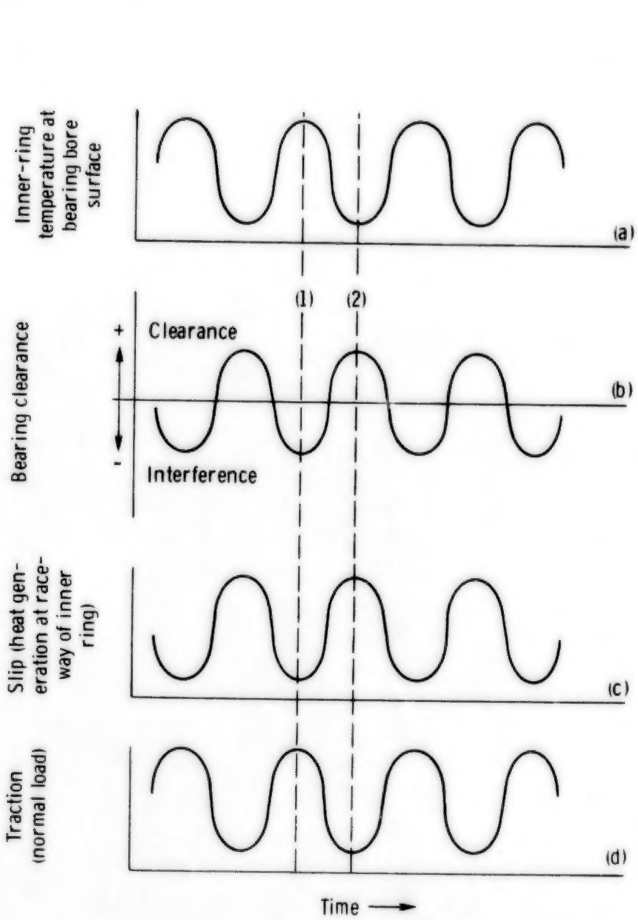


Figure 15. - Parameters affecting bearing temperature cycling with time.

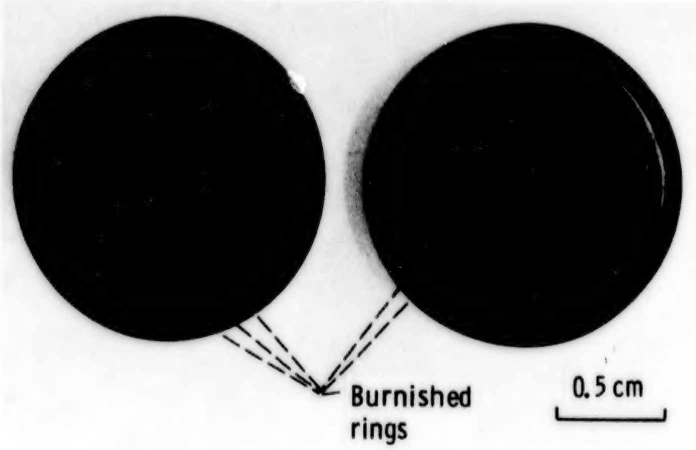


Figure 16. - Test-bearing rollers after test, showing concentric, burnished rings.

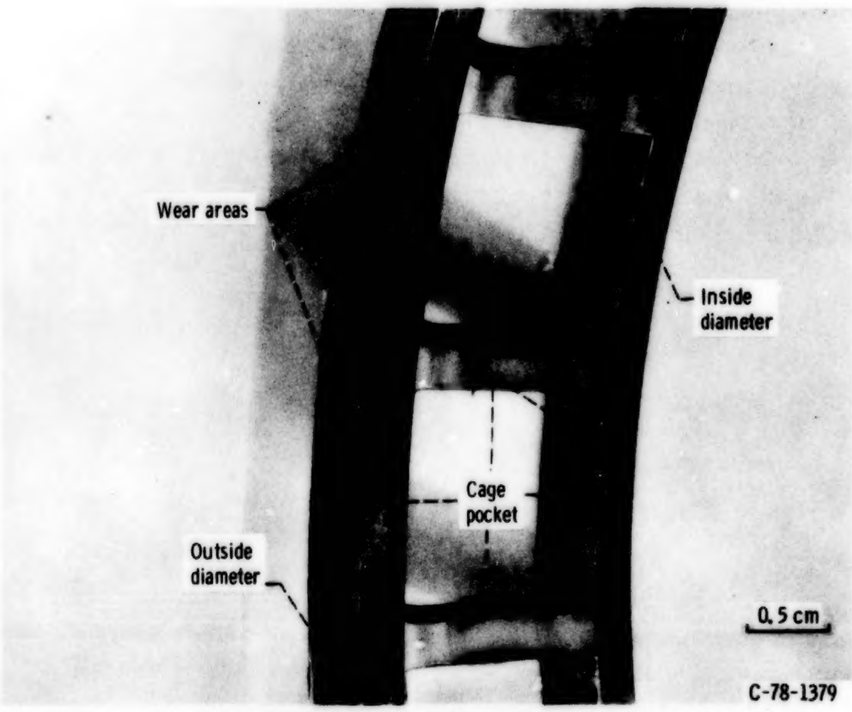


Figure 17. - Test-bearing cage after test, indicating areas of wear through silver plate.

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15. Supplementary Notes

16. Abstract

A 118-millimeter-bore roller bearing was studied parametrically at speeds from 10 000 to 25 500 rpm (1.2×10^6 to 3.0×10^6 DN). The bearing had a round outer ring (not preloaded), and provisions were made for lubrication and cooling through the inner ring. In some tests the outer ring was also cooled. The bearing ran successfully at 3.0×10^6 DN with very small evidence of cage slip. Load, which was varied from 2200 to 8900 newtons (500 to 2000 lb), had no effect on bearing temperature or cage slip over the speed range tested. Bearing temperature varied inversely with cage slip for all test conditions. Cooling the outer ring decreased its temperature but increased the inner-ring temperature. Heat rejected to the lubricant (power loss within the bearing) increased with both shaft speed and total oil flow rate to the inner ring.

17. Key Words (Suggested by Author(s))	18. Distribution Statement
Roller bearing; Rolling element bearing; Large-bore, high DN bearing; High-speed, shaft-lubricated bearing	Unclassified - unlimited STAR Category 37

19. Security Classif. (of this report)	20. Security Classif. (of this page)	21. No. of Pages	22. Price*
Unclassified	Unclassified	31	A03

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